average temperatures on the lining friction surface were consistently 40–50°C lower for friction material B; a result which has significance in the interface temperature: time: μ relationship. Some of the improved in-stop performance associated with friction material B therefore may well be associated with this reduction in interface temperatures.

8 Conclusions

8.1 Realistic simulation of braking friction using finite element modelling techniques has enabled the effects of energy transformation at the friction interface of a drum brake to be studied. Results indicate that observed characteristics of brake performance, in particular in-stop torque characteristics of brake performance, in particular in-stop torque characteristics and speed-related torque variation, are mainly governed by the combined effects of the time: interface temperature: μ profiles during braking. Thermal and mechanical distortions of the brake drum, lining or shoe are responsible for a maximum of about 15 percent of the total in-stop torque variation measured; 85 percent is due to the lining surface temperature: μ characteristic of the friction material.

8.2 Predicted results show that lining friction surface temperatures peak immediately after the commencement of a brake application and then decrease. It is this time: temperature characteristic which enables in-stop brake torque to be directly related to lining μ: temperature characteristics. In comparison, drum friction surface temperatures follow the conventional temperature: time profile and peak half-way through the brake application; a result which cannot be reconciled with observed in-stop torque variation.

8.3 Lining friction surface temperatures depend upon the thermophysical properties of the friction material. For individual brake applications where the same total kinetic energy is to be dissipated as frictional heat energy, measured bulk drum temperatures remain sensibly unaltered by the choice of friction material. However friction interface temperatures may be significantly changed by the use of alternative brake lining materials. Where this change produces a reduction in interface temperature by increased heat transfer from the friction interface, an improvement in the in-stop and speed related torque characteristics of the brake would be expected.

8.4 The effects of thermal and mechanical distortion, and those of interface temperature and pressure distributions on in-stop brake performance are greatest on large, high factor, Commercial Vehicle drum brakes. Other design of brake which have low factors (e.g., cam brakes and disk brakes) are much less affected.

8.5 The analyses so far have not attempted to include every possible effect in fine detail; the objective has been to demonstrate the principles of the effect of temperature and the rate of energy transformation at the brake friction interface on brake performance. For example, the effects of "banding" or "strip braking" across the width of the rubbing path have not been considered. However, the correlation between experimental and predicted results is sufficiently good to encourage much further work in this important area of brake technology.

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DISCUSSION

F. E. Kennedy, Jr.[1]

This interesting paper points out very clearly the important role played by surface temperature in the tribological behavior of brake materials. By including the effect of surface temperature on friction coefficient the author was able to achieve very good predictions of frictional torque in a drum brake. The analysis technique used by the author holds good promise for future use in the design and analysis of drum brakes.

Because of my interest in this work and its future application I would like the author to clarify a few points.

1. How was wear handled in the simulation technique? It was evidently used in a way that, quite correctly, resulted in a change in contact pressure distribution. In what type of tests were the coefficients of the wear rate equation developed? Were they the same as the tests used to determine the μ-θ relationship?

2. How were the thermal characteristics of the interface (third body) materials determined? Was the interfacial thermal resistance (1000 W/m^{2}K) consistent with the thermal properties given in Table 1?

3. What thermal boundary conditions were used at the liner/drum interface? Did the restraint locations on the drum ring outer surface have any influence on the shape of the predicted pressure distribution or on thermal distortion?

4. It appears that the interface temperature: time profiles shown in Fig. 5 were developed using the original assumption of constant friction coefficient. Was a modified temperature:time relationship developed using the later fric-

[1]Thayer School of Engineering, Dartmouth College, Hanover, NH 03755.

tion: temperature data (Table 3). Was conclusion 8.2 affected by the variation of friction with temperature?

5. What temperature was tabulated in Table 3, bulk or surface temperature? How was it measured? Data of the type given in Table 3 could be very useful in brake design and analysis, as was shown clearly in this paper.

P. Fancher

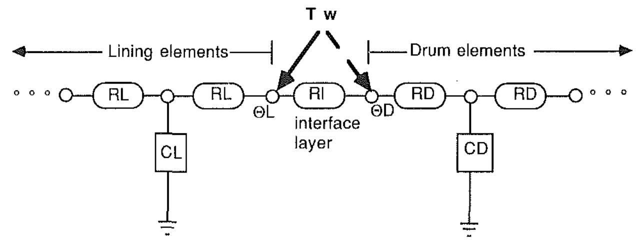
From this discusser's perspective, Dr. Day's paper provides important ideas that are pertinent to explaining the "apparent dependence of brake performance on vehicle speed." In the past, persons modeling and simulating vehicle braking have included sliding speed effects in determining the effectivenesses (torque characteristics) of brakes. Although that approach seems to work satisfactorily for stops from a given initial velocity, it appears that sometimes these "speed" influences change as a function of the initial velocity of the stop. This seemingly illogical situation is disconcerting. It implies that there exists little-known (or yet to be understood) principles of physics involving the interactions of temperature and sliding speed on friction. Dr. Day's findings indicate that much of the in-stop variation in brake torque can be attributed to the influence of the temperature of the rubbing surface of the lining on the friction coefficient between the lining and the drum (or an interfacial film deposited on the drum).

The discussions and references presented in the paper indicate that Dr. Day has developed a very detailed simulation of a drum brake. This simulation includes features that account for the influences of the mechanical, thermal, and wear phenomena occurring during a single stop. Suffice it to say that this discusser does not know of any more complete simulation of the drum brake than the one described in the paper.

The results presented in the paper show that this sophisticated model is not a reasonable predictor of brake torque for stops from 95 Km/hr *unless* the relationship of friction to lining surface temperature is included in the analysis. Given information on the relationship of lining friction to surface temperature, the calculation of lining surface temperature is a critical step in predicting brake torque. The following questions for the author concern the methodology for calculating lining surface temperaure. (These questions are offered in the spirit of enhancing understanding in applying the findings of this paper to (1) simplified analyses of the torque characteristics of commercial vehicle brakes, and (2) the stopping performances of vehicles equipped with mechanical friction brakes.)

A key issue in predicting lining surface temperature is how the heat flow corresponding to the braking power (that is, Tw, where T is the brake torque which is multiplied by wheel speed w) is entered into the temperature calculations. The attached sketch indicates two choices for the insertion of "Tw" into a one-dimensional, lumped-element, electric-circuit analogy for representing the calculation of brake temperatures. A basic question is: Why would the equivalent "heat flow," Tw, be inserted on the lining side of the interface layer? I realize that this choice is essential for the temperature results obtained in the paper, but based on physical reasoning, one might hypothesize that, because the drum is in motion, work is being done at the drum surface and the heat flow would be at the drum, as shown by the dashed line in the sketch. Possibly, the heat flow takes place on the inside surface of an interface layer attached to and rotating with the drum?

With regard to the details of the temperature calculation, what would the author estimate for the number of thermal elements needed to make a "reasonably" accurate (first-



RI – Thermal contact resistance of the interface layer
RL – Thermal resistance of 1/2 of a lining element
CL – Thermal capacity of a lining element
RD – Thermal resistance of 1/2 of a drum element
CD – Thermal capacity of a drum element
ΘL – Lining surface temperature
ΘD – Drum surface temperature
T w – "heat flow input"

Fig. 7

order) prediction of lining surface temperature? For example, past experience has shown that approximately 10 to 15 elements for the drum have appeared to be adequate for predicting drum temperatures. Why were several hundred elements included in the author's calculations? How does this number relate to the low thermal diffusivity of the friction material?

Would the author explain the apparent differences between the surface temperatures shown in Figs. 4 and 5? In particular, the maximum differences between lining and drum temperatures are very large per the results given in Fig. 5. Clearly, the large lining surface temperature is needed to obtain good correspondence between measured and calculated brake torque.

Finally, even though friction material B operates at lower temperatures, the friction of this material increases with temperature. Since these two effects tend to offset each other, the statement concerning improved in-stop performance due to lower lining temperatures is not as obvious as it would be for a friction material that loses frictional capability as temperature increases.

Author's Closure

The author would like to thank both contributors for their interesting comments, and welcomes the opportunity to discuss the work further. In answer to the specific points raised by Professor Kennedy, the wear rate equation (for material A) was developed from small sample tests (Chase machine) which also provided the μ:θ relationships. At each time step in the analysis the wear at points on the lining surface corresponding to the interface node positions was computed and incorporated into the mechanical analysis as cumulative initial offsets in the "Gap Force" method. The thermal characteristics of the material at the interface were assessed by studying the effect of wear debris as a static interfacial layer on interface contact thermal resistance. The value of interfacial thermal resistance of 1000W/m^{2}K (2) was consistent with the thermal properties insofar that these were all arrived at during the wear debris studies. At the lining/drum interface all nodes on the lining friction surface were connected to all nodes on the drum friction surface to simulate rotation of the drum. Because in the simulation the frictional heat energy was generated in the lining, this produced the required result of a circumferential distribution of temperature over the lining friction surface with a sensibly uniform circumferential temperature on the drum friction surface. The restraints on the drum ring, which permitted only radial displacement at 4 points spaced 90 deg apart were chosen to minimize any effect on pressure distribution or thermal distortion.

Temperature (θ) in both wear and friction coefficient data was bulk temperature of the metal mating body measured by embedded thermocouple. This was taken as a first approximation to the average friction surface temperature of the 25 mm square specimen. Frictional heating will, of course, generate a rise in actual interface temperature. This research work is still in early stages and further studies of μ:θ relationships, as well as material properties in general, are required. The effect of the variation of friction with temperature on the interface temperature: time relationship is an immediate area of further study.

The author appreciates the comments made by Mr. Fancher which put the purpose behind this research work into perspective. The assumption that the frictional heat flux (Tw) is all generated in the friction material is based upon the physical explanation of the generation of friction forces by the physical working (ploughing, shearing) of the contacting surfaces. The brake lining, being considerably softer than the metal mating surface, is expected to undergo the greater deformation and, in this study, the deformation of the metal surface is considered to be negligible in comparison with that of the lining. It has been observed that transfer films on the friction surface of brake drums (and disks) can affect heat flow in brakes, but in general such films are very thin. More significant is the wear debris which acts as a third body between lining and drum or rotor, but this is usually associated with the lining and considered stationary rather than moving with the drum. A temperature calculation of the linear system model type suggested by Mr. Fancher could indeed be used for brake temperature prediction, requiring far fewer elements than needed for the finite element analysis. Convergence and stability of finite element transient temperature solutions can present significant problems with the analysis of heat transfer by conduction through low thermal diffusivity materials such as brake linings. In this case large numbers of thermal elements are necessary to obtain a good solution. The complexity of the analysis presented has, however, been necessary to establish the principles and validity of the simulation. The apparent differences between Figs. 4 and 5 again relate to the complexity of the model; Fig. 4 shows actual temperatures taken through a radial section of lining and drum at the end of the brake application as in Fig. 3. Different radial sections show different temperature profiles which will, of course, alter with time. Ideally a local value of friction coefficient would be applied to each point in the friction interface, resulting in a circumferential variation in μ. Although this is within the capabilities of the analysis, the simpler approach of an average surface temperature over the lining friction surface and no circumferential variation in μ was found to yield good results as illustrated. Figure 5 therefore shows lining friction surface temperatures averaged over the lining arc length for each point in time. The characteristic form of this average lining surface temperature: time relationship is maintained at different positions along the lining arc but actual surface temperature: time results show some variation in magnitude. The final comment regarding the combined effect of temperature and μ:θ relationship highlights a result of this research which has interesting implications for brake design engineers.